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DSTR/501-M62B DYNAMIC INTERFACE CRITICAL SPEED PROBLEM

Detroit Diesel Allison
4700 West 10th Street
Indianapolis, Indiana 46206

May 1977

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Approved for public release;
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Prepared for

EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
Fort Eustis, Va. 23604

EUSTIS DIRECTORATE POSITION STATEMENT

This report provides detailed insight into a specific R&D problem experienced during the Dynamic System Test Rig (DSTR) phase of the HLH program. A discussion of the evolution of the problem, details of the subsequent design investigation, and the resulting solution are presented. Results of this contract are being integrated with other R&D problem identification efforts at the Eustis Directorate to establish research and development programs to improve engine/drive train/airframe dynamic interface characteristics of future Army aircraft systems.

Mr. Allen C. Royal, Propulsion Technical Area, Technology Applications Division, served as Project Engineer for this effort.

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A dynamic interface problem was encountered during the Dynamic System Test Rig (DSTR) portion of the Heavy Lift Helicopter (HLH) program. This problem involved the dynamic incompatibility of the original designs of the Detroit Diesel Allison (DDA) 501-M62B engine and the Boeing Vertol (BV) DSTR shafting. Presented is a detailed discussion of the interface problem and the steps taken toward solution. Design modifications to both the engine →

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20. ABSTRACT (continued)

and shafting were necessary to synthesize an acceptable drive train configuration. Recommendations are proposed for avoidance of engine/airframe interface problems in future programs.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

INTRODUCTION

A modern high-performance helicopter represents a meld of many complex components and subassemblies. The proper combination of these parts must be accomplished to attain high performance and reliability with low risk. A large number of technical problems must be resolved in the design and development process. One area which deserves considerable attention relates to bridging the interface between the airframe and the engine. Modern helicopter designs favor the use of drive shafting which rotates at engine power turbine rotational speed without reduction. The speed reduction for the helicopter rotors is accomplished in the helicopter transmission. Such a concept results in smaller diameter (lighter), high-speed drive trains. Shafting critical speeds become more important than in designs which use speed reduction gearboxes within the engine or close to the engine output shaft. It is essential that a coupled shafting/engine design be determined which has acceptable vibratory characteristics.

During the development period of the Heavy Lift Helicopter (HLH) program, a major dynamic compatibility problem was discovered which involved the coupling of the engine and drive shafting in the Dynamic System Test Rig (DSTR). This rig is an integrated rotor and drive shafting test vehicle for developing the advanced technology components for the HLH. Three engines transmit power to a combiner transmission (mixbox), through shafting which is rotating at engine power turbine rotational speed. Analysis of the originally proposed DSTR shafting and the originally proposed 501-M62B engine showed that each major component of the drive train had acceptable rotor dynamic characteristics when analyzed separately. A more detailed coupled shafting/engine analysis indicated that the combination was incompatible. An involved investigation ensued, and after both the shafting and the engine were modified, a compatible drive train design was accomplished.

This report presents a discussion of the evolution of the interface problem, the details of the following design investigation, and the resulting solution. A summary and set of recommendations for future programs are included.

EVOLUTION OF THE INTERFACE PROBLEM

The interface problem, resulting from the coupled dynamic behavior of the 501-M62B engine and the HLH drive train, surfaced during the DSTR development portion of the HLH contract. Although dynamic analyses of the engine showed acceptable vibratory characteristics, and dynamic analyses of the drive train indicated an acceptable design, subsequent coupled analyses revealed an interface problem. This problem evolved as the subcontract between DDA and BV progressed and communication links were established among the technical people. This section relates the evolution of this problem.

During the formative stages of the 501-M62B engine design (prior to any contract award), numerous engine configurations were explored. One such configuration is that depicted in Figure 1. This design features a front-drive free-turbine engine with an extended torquemeter assembly. The torquemeter design closely resembles that used in the T56-A-18 turboshaft engine. However, it was about this point in time that BV stated a desire to shorten the engine. The change was accomplished by including a part of the torquemeter assembly within the power turbine shafting as indicated in Figure 2. This is the preliminary engine configuration included in the 22 November 1971 DDA proposal to BV for operation in the DSTR. The design features a reduced extension of the torquemeter shaft of 8.28 inches from the engine inlet flange.

At the first coordination meeting between DDA and BV, a modification of the engine, relative to torquemeter overhang, was discussed. Based upon a more detailed engine nacelle design, BV requested that the interface be moved forward to more easily accommodate the engine air inlet duct with minimum blockage and to facilitate inspection of the flexible coupling. This change was agreed on, and the interface was moved to a point 17.21 inches from the engine front face.

The initial configuration of the airframe shafting system featured two shafts, 42.25 inches in length, three Thomas couplings, and one support bearing between the engine and combiner box. A schematic of this arrangement is shown in Figure 3. The shafting interface was made at the internal spline of the Thomas coupling adapter. The engine provided a mating external drive spline which terminated 17.21 inches forward of the engine inlet housing flange. Support of the output shaft/torquemeter configuration was provided by a machined aluminum casting which terminated 13.8 inches forward of the engine inlet flange. A layout of the modified

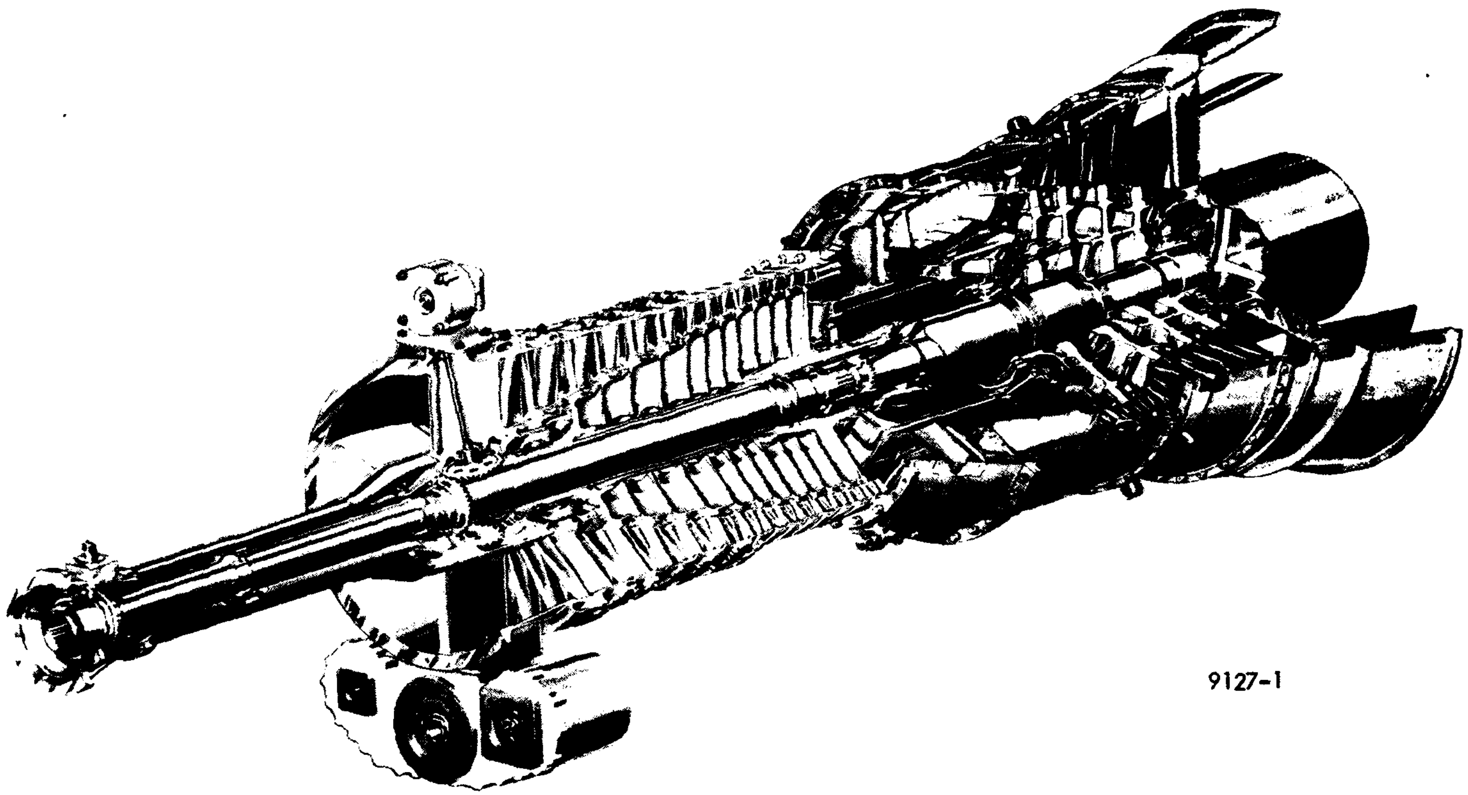

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Figure 1. Original 501-M62B Configuration with Extended Torquemeter.

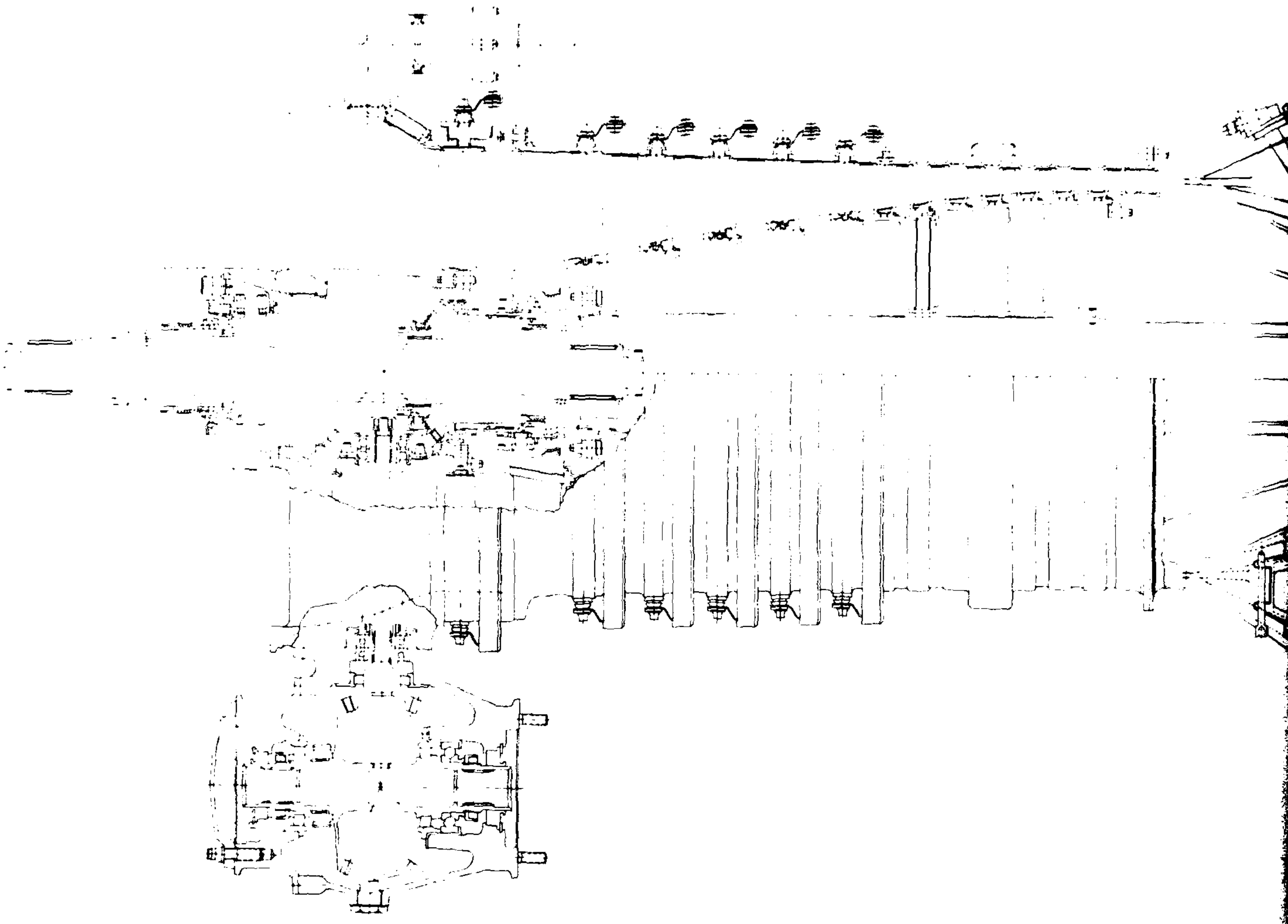
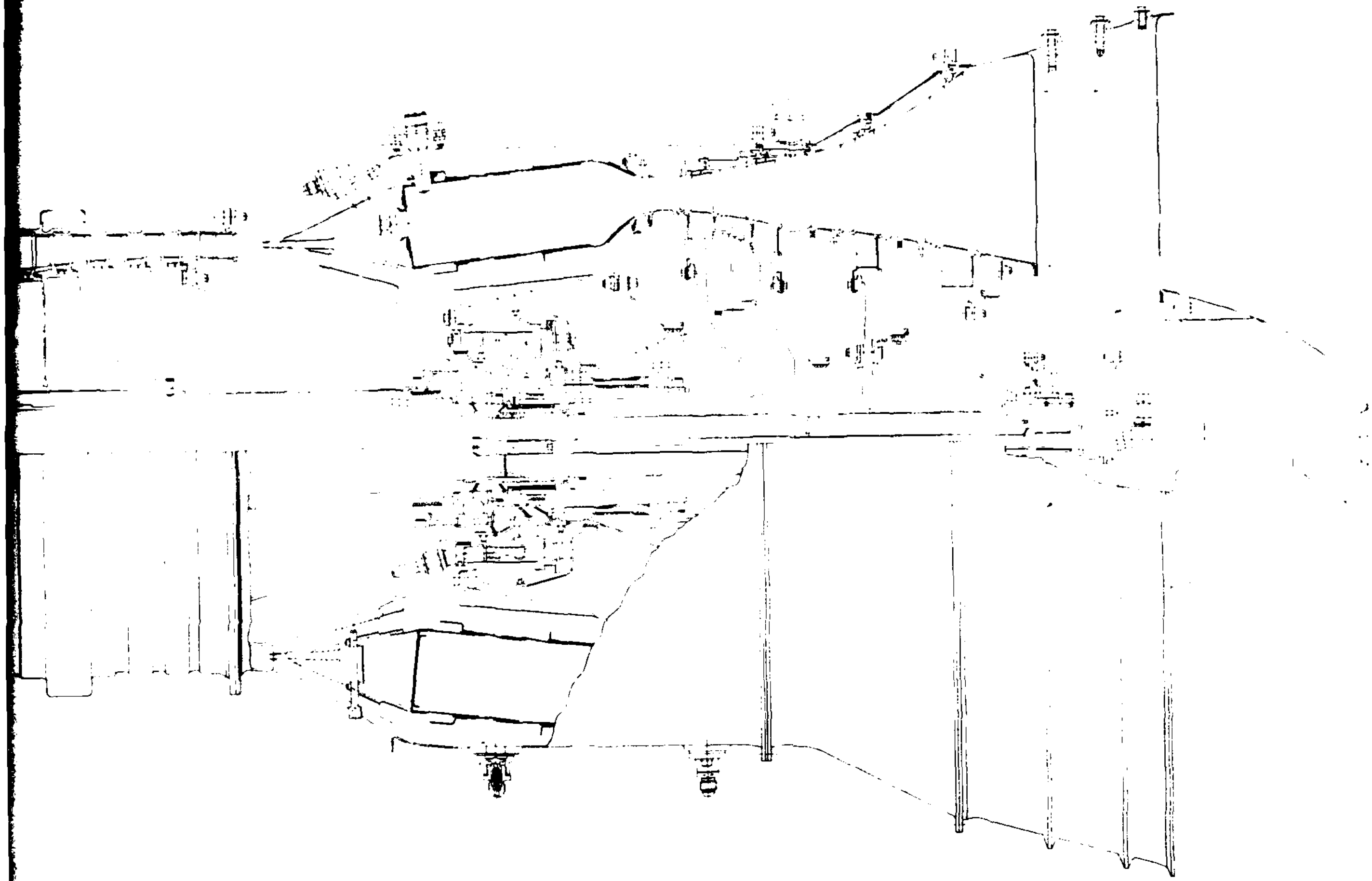


Figure 2. 501-M62B Proposal Configuration.



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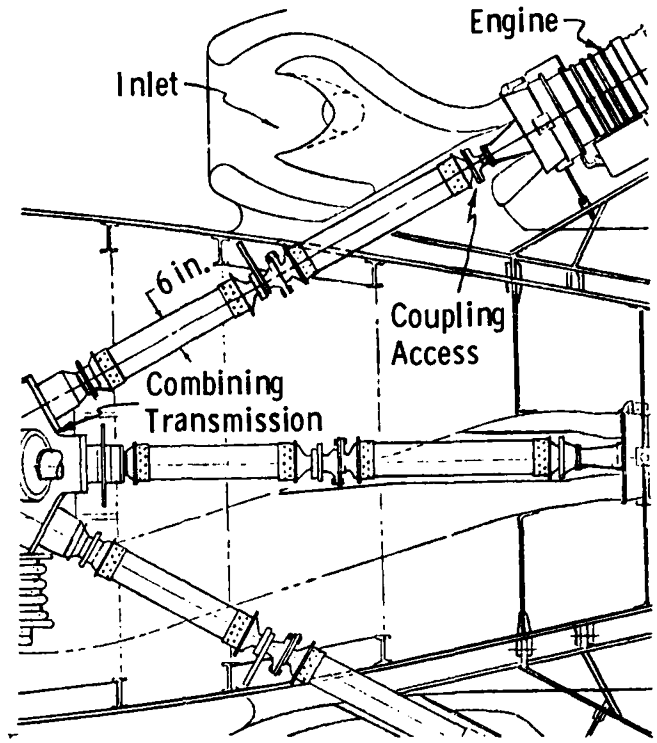


Figure 3. Schematic of Original DSTR Shafting Arrangement.

torquemeter configuration is shown in Figure 4. Structural analyses of the extended torquemeter configuration indicated the following end flexibilities:

TABLE 1. End Flexibilities of 501-M62B With Extended Torquemeter

Load	Radial Deflection (in.)	Pitch Rotation (rad)
1 lb force	8.93×10^{-6}	0.944×10^{-6}
1 in.-lb moment	0.944×10^{-6}	0.140×10^{-6}

This implies an end translational stiffness of 112,000 lb/in.

These flexibility values were made available to BV as per Military Specification MIL-E-8593A. Paragraph 3.1.2.6.2 of this specification states, "The estimated stiffness of the engine in resisting loads and moments applied at the outboard end of the output shaft, relative to the engine mounting points, shall be stated in the engine specification. The first "free-free" lateral and vertical engine bending modes shall be specified."[1]

A reduction in shaft and Thomas coupling diameter was desired to further reduce the engine inlet airflow blockage and reduce weight. A preliminary study of alternatives was performed by BV in which critical speeds of the shafting were computed for various combinations of shaft diameter and total effective engine end fixity. The model used in the computations was basically a beam element representation of the shafting with local flexibilities included to account for the Thomas couplings. The shaft/engine interface end of the shafting was considered tied to ground with a translational stiffness equal to the engine end stiffness. The mixbox was modeled at the other shaft end. Results of this study are shown in Figure 5. It shows that an extremely high stiffness would be needed at the engine for a 5.0-inch-diameter shaft to have a sufficient margin above the 11,500 rpm design speed. A 7.27-inch-diameter shaft has acceptable critical speed margin with the original estimate of end stiffness, but this large diameter exhibits unacceptable inlet blockage. The compromise diameter of 6.0 inches would require an engine end stiffness increase from 112,000 lb/in. to 150,000 lb/in. This would increase the critical speed margin to 20 percent above the design speed. DDA agreed to modify the engine to achieve the desired end stiffness of 150,000 lb/in. by changing the torquemeter housing from aluminum to steel. A subsequent analysis with revised shaft adapters, Thomas coupling size, and shaft tube length; improved mixbox idealization, and 150,000 lb/in. effective end stiffness revealed that the critical speed was actually 15,300 rpm, or 33 percent above the design speed. The resulting critical speeds and associated mode shapes are shown in Figure 6. Although it is not entirely obvious from the mode shapes, the second shafting critical speed is the first mode which has significant participation of the torquemeter. This mode incorporates substantial whirl of the torquemeter housing and some shaft bending, and is very sensitive to output shaft coupling weight and unbalance. This is the mode with which the remainder of these discussions

[1] Military Specification, ENGINES, AIRCRAFT, TURBOSHAFT AND TURBOPROP, GENERAL SPECIFICATION FOR, MIL-E-8593A, U.S. Government Printing Office, Washington, D.C., 15 October, p. 6.

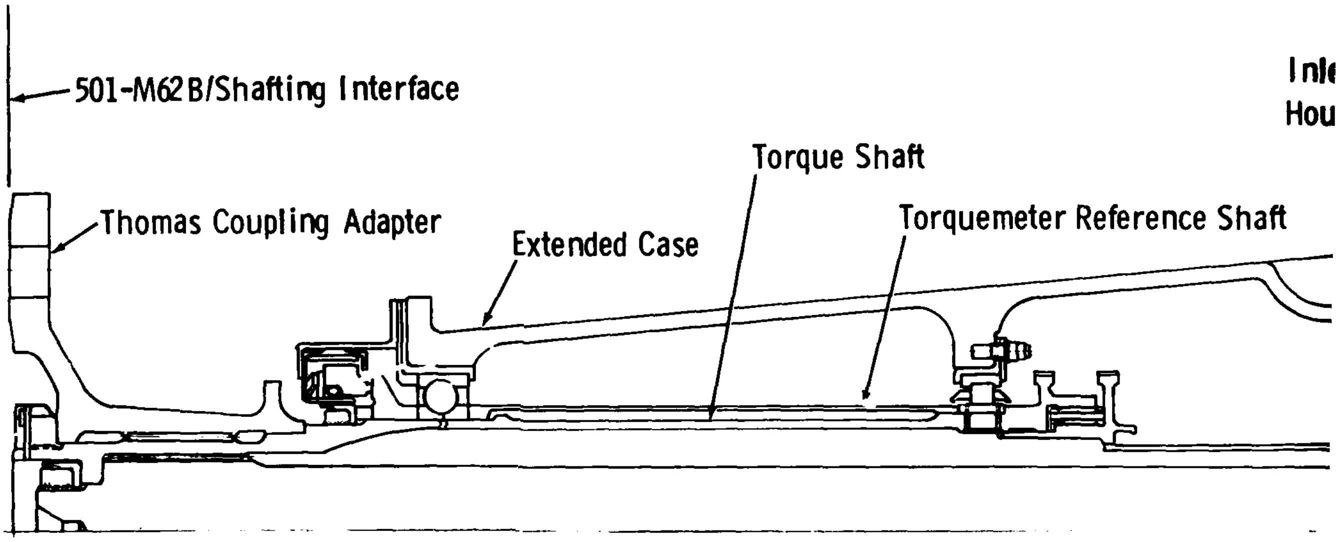
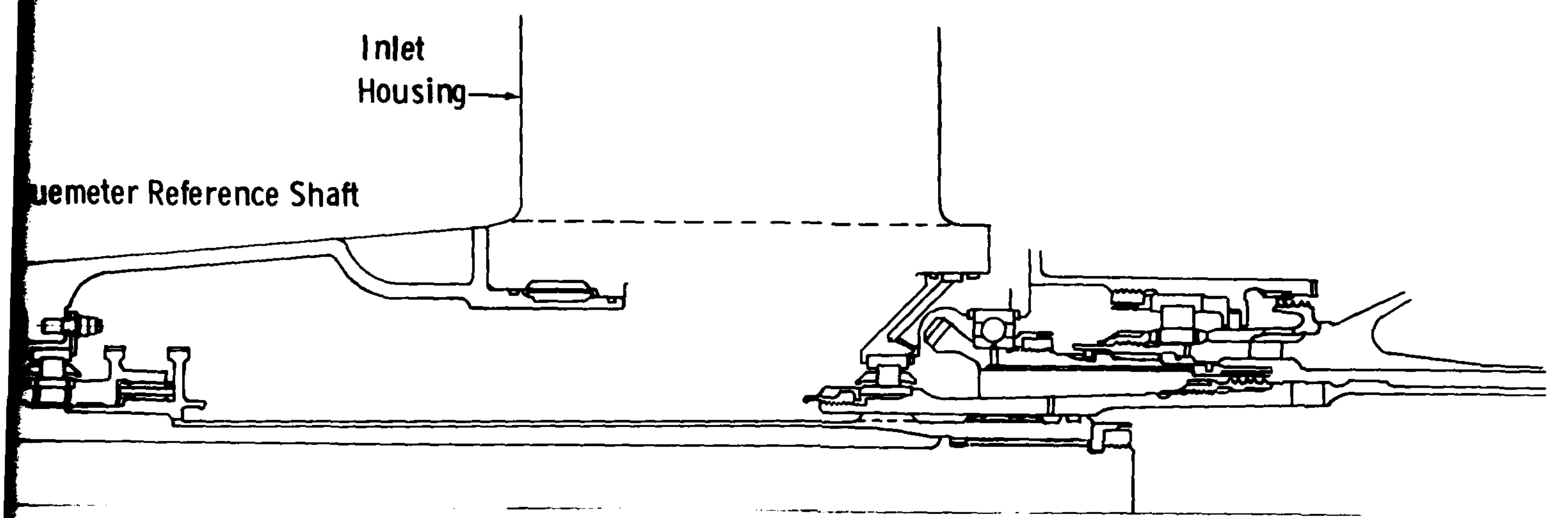


Figure 4. Layout of Modified 501-M62B Extended Torquemeter Configuration.



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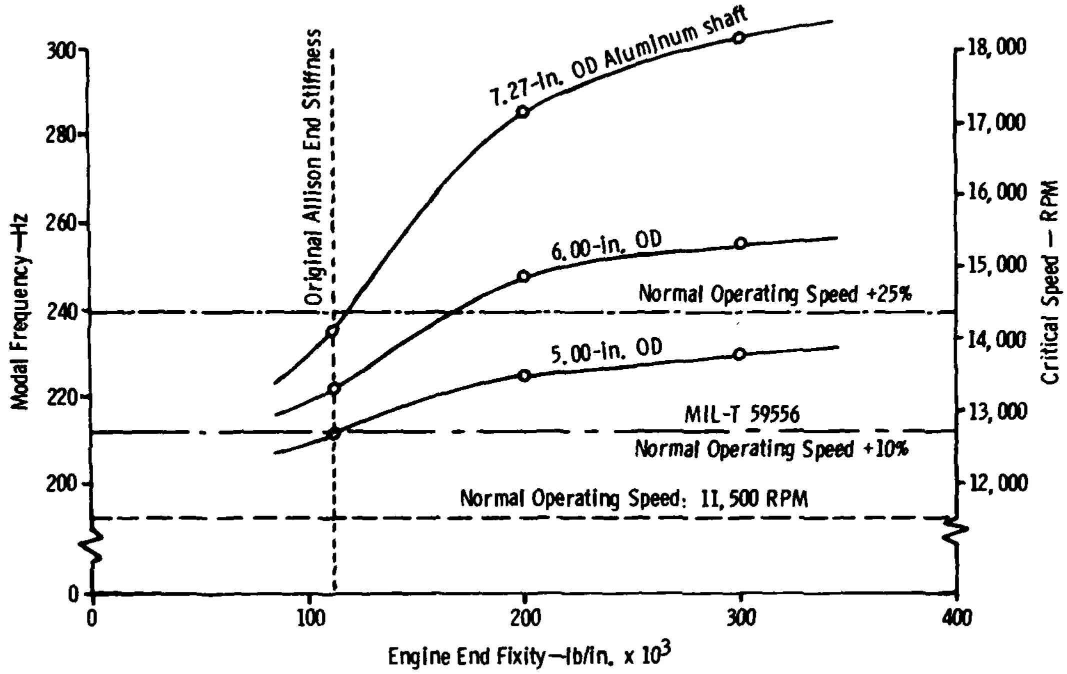


Figure 5. DSTR Shafting Trade-off Study Between Shaft Diameter and Engine End Stiffness.

are concerned. The true character of this mode will become evident as the discussions continue. It is important to note that an analysis of the shafting system was performed using engine data required by Military Specification MIL-E-8593A. The analysis indicated an acceptable configuration.

Due to a strong dependence of the shaft critical speeds on the engine end stiffness, it was decided that DDA should perform a coupled analysis using a sophisticated torquemeter model and couple that with a model of the BV shafting. In that way the engine end fixity could be modeled along with the torquemeter mass effects. A schematic of the torquemeter model is shown in Figure 7. The analysis showed a conical whirl (ω) of the torquemeter shaft and housing at 13,600 rpm for torquemeter shaft bearing support rates of 1×10^6 lb/in. The mode drops to 12,700 rpm for conservative bearing support stiffness estimates of 500,000 lb/in. These results

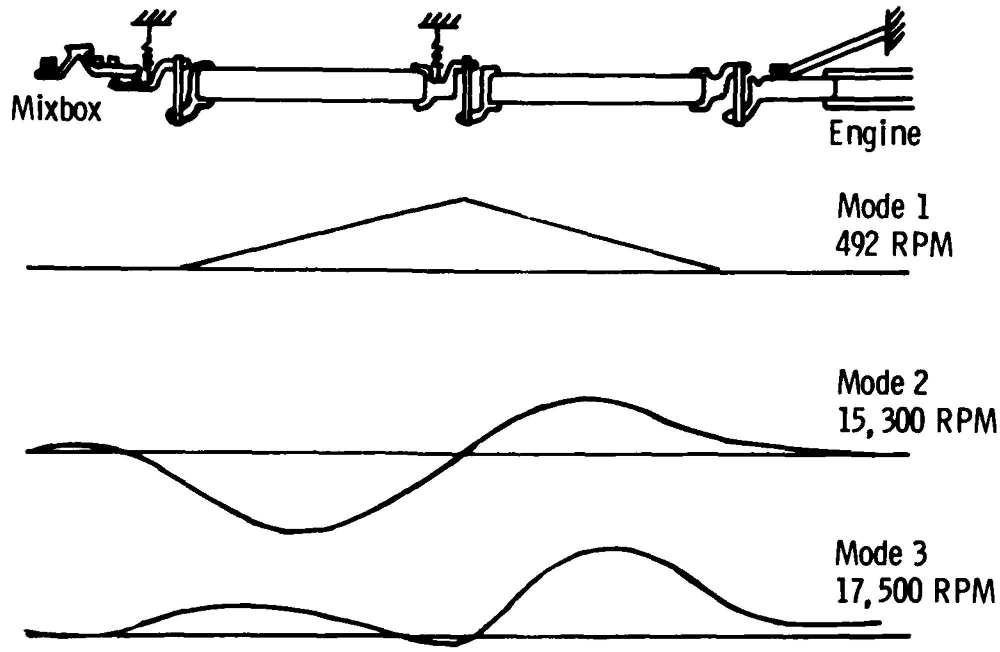
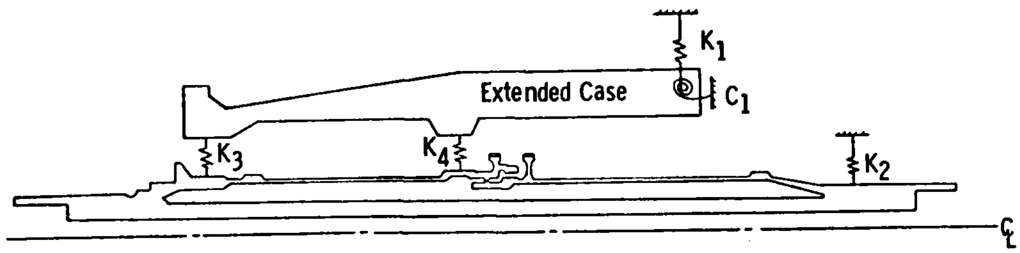


Figure 6. Critical Speed Analysis Results for Initial 6-Inch-Diameter Shafting Configuration—End Stiffness 150,000 Lb/In.

are summarized in Figure 7. Since the design speed was 11,500 rpm, the results of this analysis indicated an unacceptable drive train when coupled with the engine.

Results of the DDA analysis were subsequently substantiated by a BV analysis. Figure 8 shows the equivalent finite element model of the torquemeter used by BV. A whirl frequency of 13,566 rpm was computed as compared to 13,598 rpm in the DDA analysis. The interface problem, that of insufficient critical speed margin, was established. Further analysis and/or testing, leading to a viable drive train configuration, was required. The discussion presented in this section has shown the evolution of a dynamic interface problem. An early analysis of the drive train using end fixities, required in Military Specification MIL-E-8593A, indicated an acceptable design. A coupled shafting/engine torquemeter analysis, with a more refined model of the torquemeter shaft and housing, indicated an unacceptable design. The problem was identified.



	Case 1	Case 2	
C_1	140×10^6	140×10^6	lb-in./rad
K_1	1×10^6	1×10^6	lb/in.
K_2	500,000	500,000	lb/in.
K_3	1×10^6	500,000	lb/in.
K_4	1×10^6	500,000	lb/in.
•ω	13,598	12,715	CPM

•Includes B/V Shafting

Figure 7. 501-M62B Torquemeter Schematic.

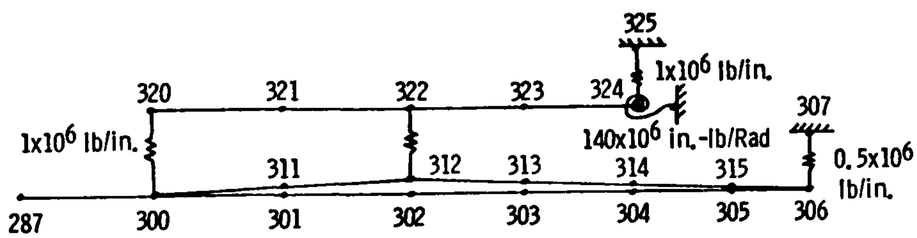


Figure 8. Equivalent Torquemeter Model Used in BV Analysis.

DESIGN INVESTIGATIONS

Once the interface problem was determined, it became necessary to effect a solution which could be readily assimilated into the program with minimum cost and delay. Design investigations were initiated at both DDA and BV in search of a viable configuration. This section presents a discussion of these investigations.

The discovery of a dynamic interface problem was made after engine hardware was committed. The engine inlet frame and torquemeter extended case hardware was available. A static deflection test was ordered to verify some of the spring rates which were estimated and used in the foregoing analyses. In addition, a finite element idealization of the structure was performed for correlation purposes and to serve as a vehicle for evaluating structural design changes.

A sketch of the test setup is shown in Figure 9. Loads were applied using hydraulic rams at incremented load levels through 1500 pounds. Figure 10 indicates the deflections noted for a 1500-pound load applied at the forward flange of the housing. The resulting spring rate was determined to be 156,000 lb/in. The finite element model depicted in Figure 11 was used to compute the spring rate analytically. A rate of 167,000 lb/in. was computed, showing very good correlation. The finite element model was deemed sufficiently accurate to be used for further analysis. The original estimates of the engine end stiffness were based on front and rear torquemeter bearing spring rate estimates of 650,000 lb/in. and 750,000 lb/in., respectively. A subsequent analysis of these bearings, using a computer program provided by SKF Industries, Incorporated, indicated the rates to be closer to 1.6×10^6 lb/in. for the load range anticipated at the bearings. The newer bearing rates were incorporated in the analysis. Using the finite element model of the engine inlet housing and torquemeter housing, and coupling the torquemeter shaft through bearing flexibilities, the effective end stiffness translational spring rates were computed for the configurations listed in Table 2.

It can be seen from the configurations investigated that various methods for stiffening the engine extended torquemeter were under consideration. These included:

- Stiffening the inlet housing
- Shortening the housing extension
- Incorporating struts between the inlet housing and torquemeter housing—this design alternative was carried along as a possible interim solution since the use of struts in the inlet flow would cause blockage and inlet distortion and could not be considered an acceptable long-range design solution.

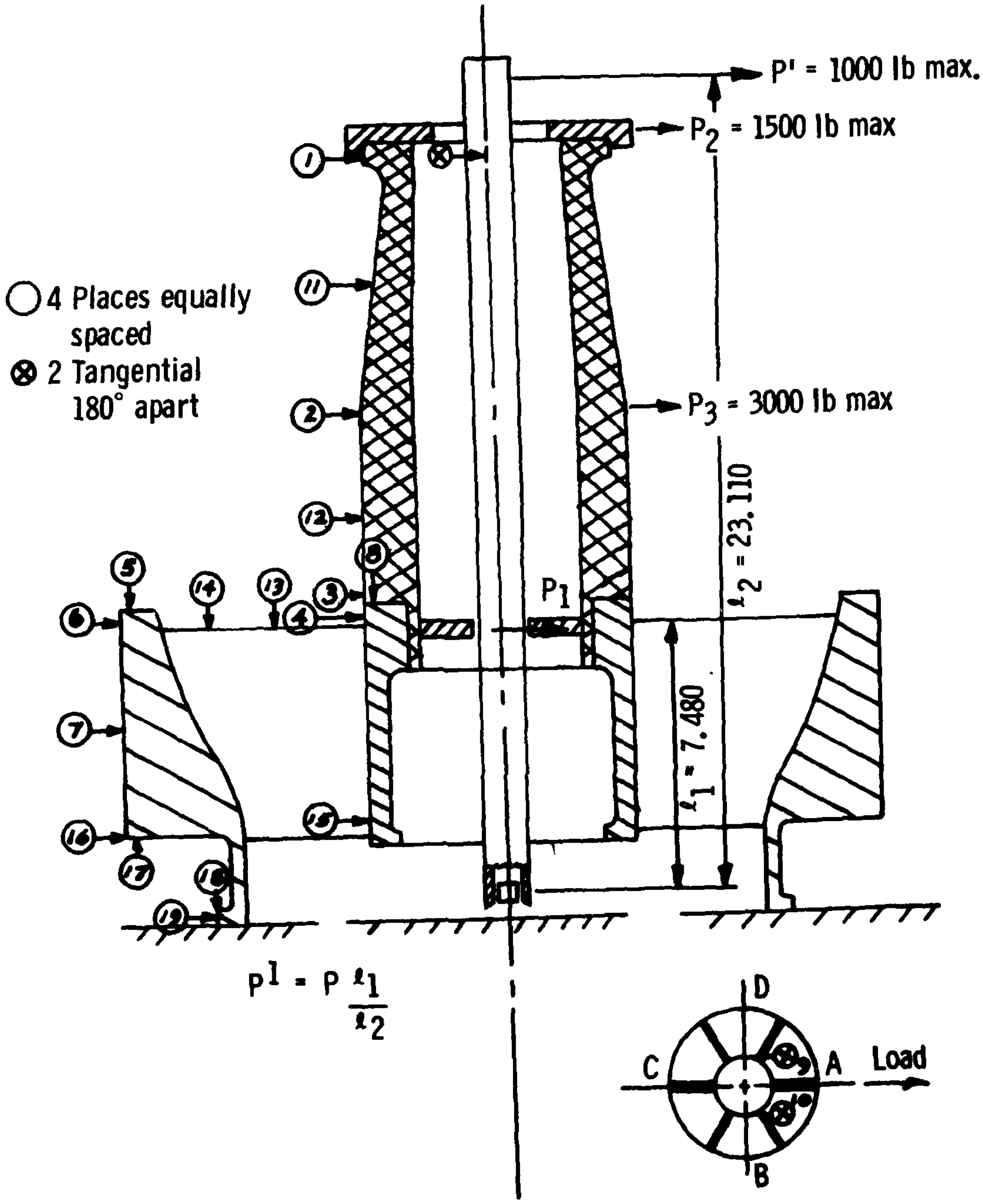


Figure 9. Test Setup for Torquemeter Housing Static Load Test.

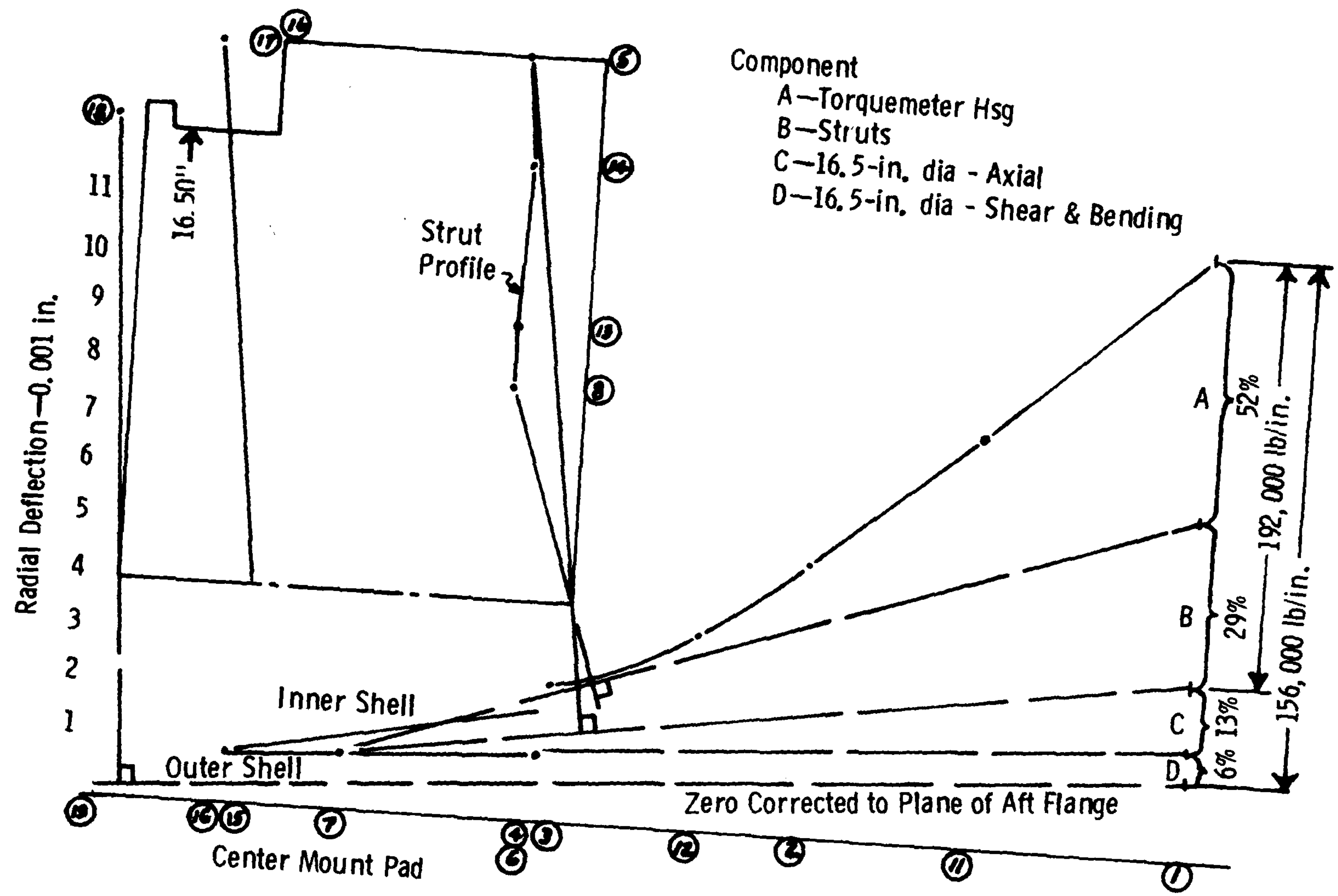


Figure 10. Deflections Resulting From 1500-Pound Load in Vertical Plane.

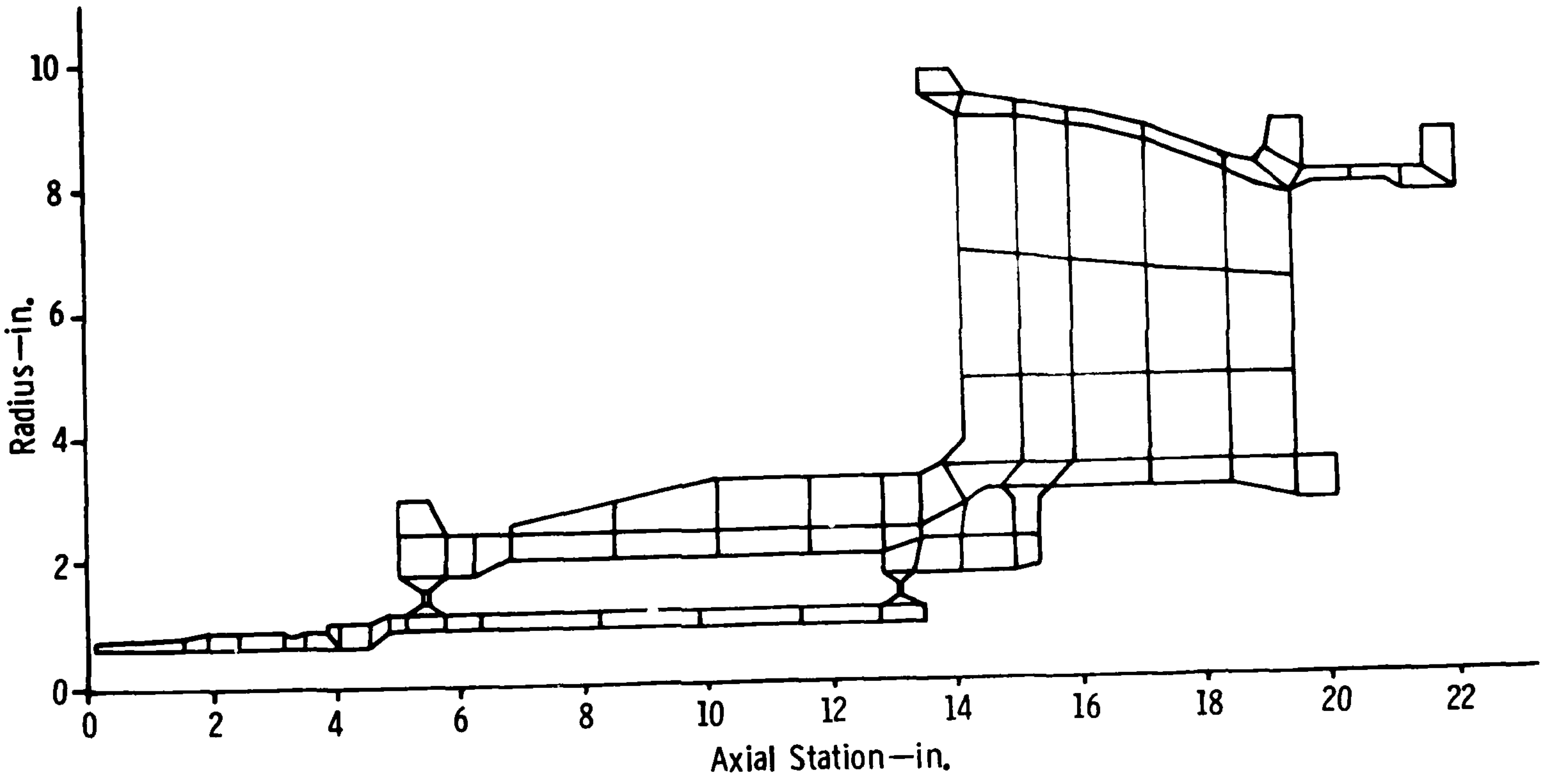


Figure 11. Finite Element Model of Engine Inlet and Torquemeter Housing.

TABLE 2. Computed Engine End Translational Rate

Configuration	Spring Rate (lb/in.)	Deflection per 10,000 lb (in.)
1. Current released design (650,000 and 750,000 lb/in. bearing stiffness)	61,000	0.164
2. Same as 1, with oil holes removed	70,000	0.142
3. Same as 1, using a bearing stiffness of 1.6×10^6 lb/in.	72,000	0.139
4. Same as 3, with oil holes removed and output shaft stiffened	88,500	0.113
5. Same as 4, with inlet housing stiffened	109,000	0.092
6. 4 inch shorter torquemeter housing	160,000	0.062
7. Same as 4, with six 0.8-in.-diameter aluminum struts, pin jointed between the inlet housing and the front of the torquemeter housing	176,000	0.057
8. Same as 7, with struts attached to torquemeter housing at inner bearing plane	176,000	0.057
9. Same as 8, except with steel struts and solid torquemeter housing between bearings	208,000	0.048

Based on the experimental test data and finite element analyses, changes were made to the dynamic rotor model. The rotational deflection of the inlet housing had previously been represented by a clock spring having a rate of 140×10^6 in.-lb/radian. A rate of 75×10^6 in.-lb/radian is more appropriate. With the redefined dynamic model, the critical speed occurs at 9,235 rpm. Using this same model but with an infinitely rigid torquemeter housing, the critical speed increases to 15,246 rpm, a margin of 32 percent above the normal operating speed. The significance of this analysis points to the fact that a satisfactory solution to the problem likely requires some change to the DSTR shafting as well as to the engine.

The question of what design margin to use in the subsequent design investigations was discussed. It was mutually agreed by DDA and BV that the first coupled system critical speed above normal design speed (11,500 rpm) should have at least a 25-percent margin, or occur above 14,375 rpm. Since the original shafting/engine configuration does not satisfy this criterion, numerous design configurations were analyzed to determine their critical speeds.

Three extended torquemeter configurations and one shafting-contained torquemeter configuration were considered. These were:

1. Original TM—This was the proposed torquemeter configuration shown in Figure 4.

2. Short TM—This intermediate configuration was described only by sketch and represents a 4-inch shortened torquemeter configuration.

3. New Short TM—This configuration is a stiffened version of the Short TM.

4. Shafting-Contained TM—In this configuration the torquemeter is contained in the center section of a three-section BV shafting arrangement.

These torquemeters were combined with two basic shafting configurations:

1. Two-section shaft—This shafting configuration was proposed with the original torquemeter and is shown in Figure 3.

2. Three-section shaft—This shafting configuration is characterized as having three equal 6-inch-diameter shafting lengths separated by Thomas couplings. A sketch of the design is shown in Figure 12.

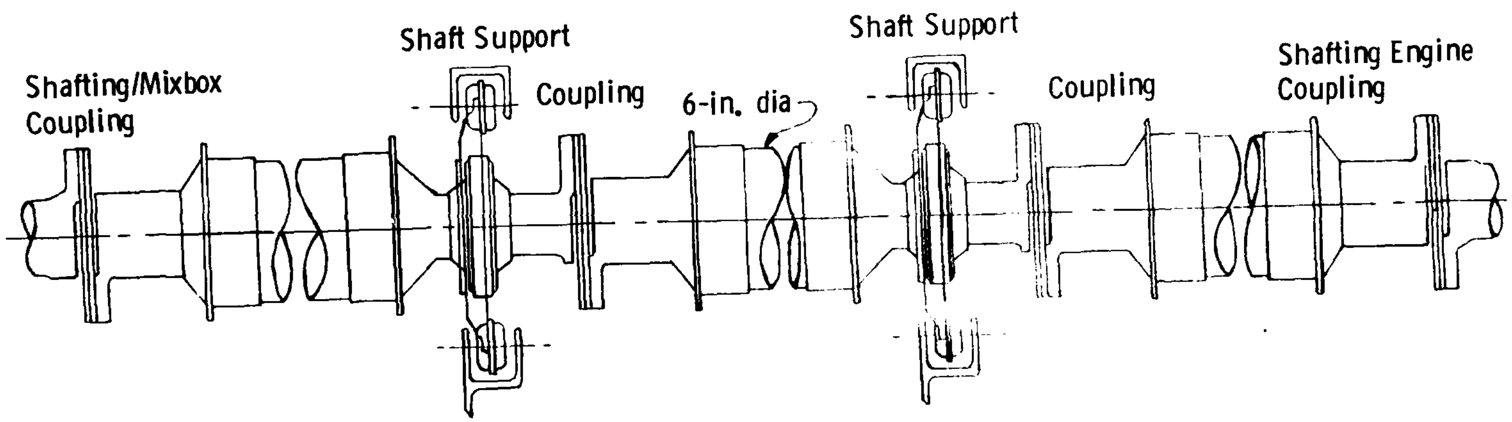


Figure 12. Sketch of Three-Section DSTR Shafting Arrangement.

Two models were used in the analysis:

1. W/O Engine—This model is that previously defined and includes models of the BV shafting and DDA torquemeter. Analyses using this model were eventually halted in favor of the shafting/engine coupled model.

2. W/Engine—This model couples the models of the BV shafting and the entire engine, including high pressure rotor, low pressure rotor, torquemeter, and casing.

Table 3 shows a composite listing of the results of early analyses of these design combinations. As can be seen, no non-strut configuration has a critical speed in excess of the 14,375 rpm goal when the coupled shafting/engine analysis is used. The most promising design not employing struts appears to be the shortened torquemeter with three sections of shafting. It is interesting to note that this design has an effective stiffness of 167,000 lb/in. and still does not provide an adequate dynamic margin. The inadequacy of the shafting/torquemeter analysis is evident from the large discrepancy of the results. This is due primarily to the fact that the critical mode involves substantial coupling of the LP turbine. Consequently, the more refined shafting/engine model was used in subsequent analyses.

Further variations of the new short torquemeter and three-section shafting were made to determine new trade-offs. The basic approach was to determine the effects of various engine stiffening schemes. Table 4 summarizes the calculated shaft/engine critical speed for the design variations indicated. From the analysis of these results and those shown in Table 3, it becomes clear that to obtain an acceptable non-strut configuration it is necessary to:

- Move the Thomas coupling as close to the engine as possible.
- Maximize the engine inlet/torquemeter housing moment stiffness.
- Reduce the effective overhung weight supported by the torquemeter.

One approach which offered some promise was to include the torquemeter as part of the BV shafting. That allowed the engine/shafting interface to be moved as close to the engine as possible. A sketch of one of the designs considered is shown in Figure 13. This shows a three-section shaft with the torquemeter contained in the center section and a Thomas coupling overhang of 3.03 inches from the engine inlet. Analyses of this configuration were performed as a function of the amount of Thomas coupling overhang. One analysis showing promise featured an overhang of 2.4 inches.

TABLE 3. Composite Listing of Early Design Study Results

Configuration Analyzed	No Struts		Aluminum Struts		Steel Struts	
	W/O Engine	W/Engine	W/O Engine	W/Engine	W/O Engine	W/Engine
Original TM 2-Section Shaft	9,235 (71,500)	10,157	13,112 (161,500)		14,000 (208,000)	
Original TM 3-Section Shaft	9,713 (71,500)		14,165 (162,500)		15,500 (208,000)	13,223 (104 Mils)
Short TM 2-Section Shaft	12,918 (150,000)		15,160 (258,000)			
Short TM 3-Section Shaft	14,243 (147,000)	12,425	18,200 (258,000)			
Short TM 3-Section Shaft - Case Modulus × 9		14,265				
New Short TM 3-Section Shaft	14,200 (167,000)	12,400 (50 Mils)			18,300	14,055
New Short TM 3-Section Shaft 12-in. First Section	15,500 (167,000)	13,082				

Numbers in parenthesis W/O engine refer to effective spring rate.

Numbers in parenthesis W/engine refer to double amplitude response at the Thomas coupling nearest the engine to 1.0 ounce-inch of unbalance at that coupling.

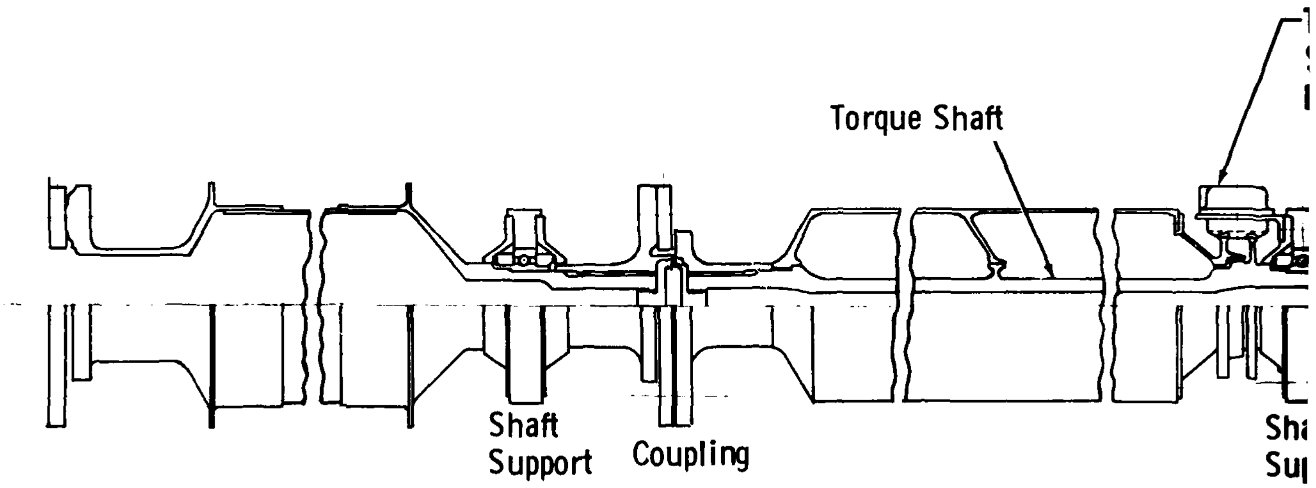
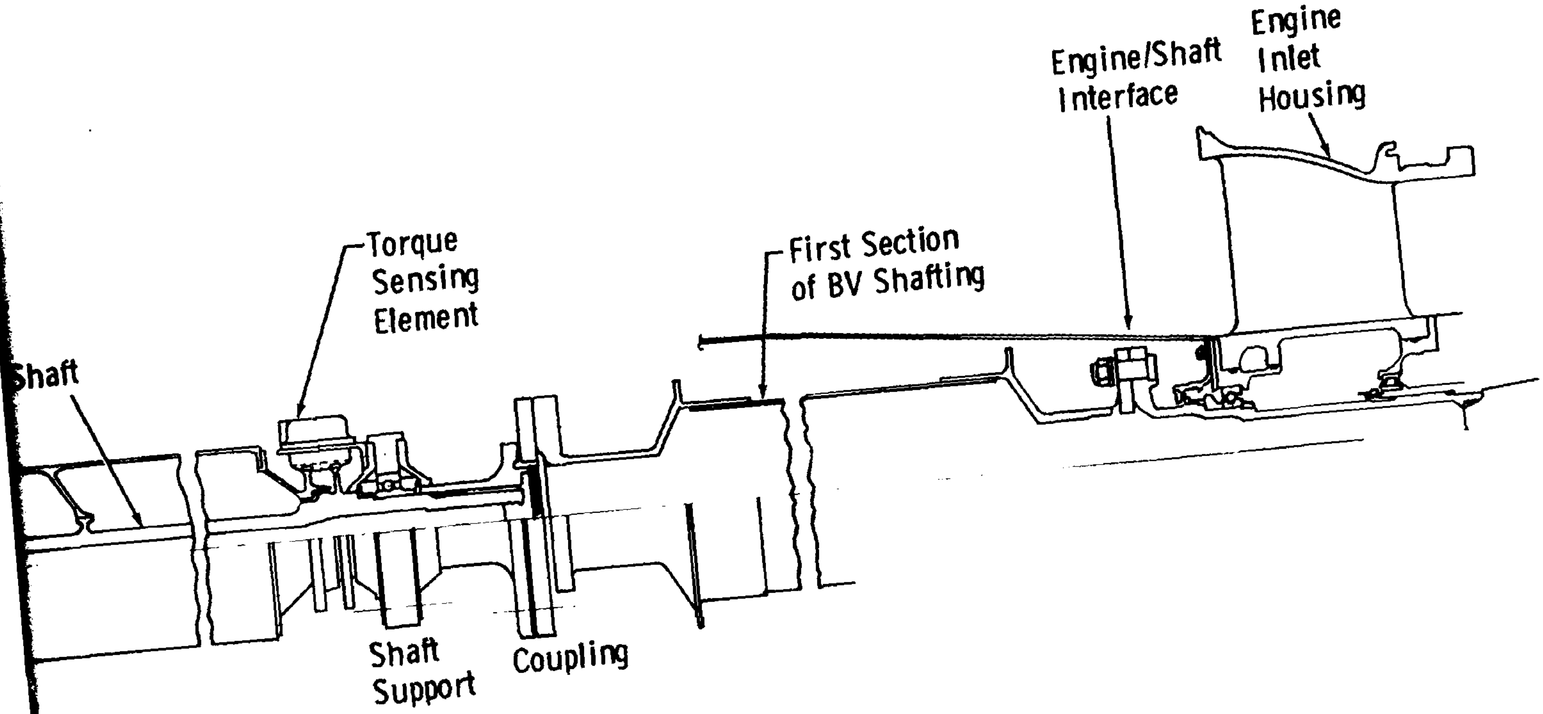


Figure 13. Layout of Shaft-Contained Torquemeter Configuration.



Engine
Inlet
Housing
Engine/Shaft
Interface
Torque
Sensing
Element
First Section
of BV Shafting
Shaft
Shaft
Support
Coupling

uration.

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TABLE 4. Critical Speeds of Three-Section Shafting With Various Engine Stiffening

Configuration	Critical Speed of First Shaft Mode (rpm)	Δrpm From Nominal
Nominal	12,452	--
Steel Compressor Case	12,929	477
Gusseted Compressor Case	12,560	108
Half Thomas Coupling Weight	13,460	1,008
Coupling 2 in. Closer to Engine	13,046	594
Coupling 3 in. Closer to Engine	13,217	765
Clock Spring (C_1) 90×10^6	12,452	--
100×10^6	12,503	51
150×10^6	12,659	107
200×10^6	12,735	283
Front Engine Mount		
Moment Fixity 0	12,452	--
1×10^6	12,460	8
5×10^6	12,491	39
10×10^6	12,528	76
50×10^6	12,790	338
100×10^6	13,044	592

A geometric representation of the analytical model of the engine is shown in Figure 14 for reference. This sketch is presented to define the axial datum. The calculated critical speeds and associated mode shapes are shown in Figure 15. The first three modes are basically rigid body modes of the shafting and engine. The fourth mode is one of HP rotor conical whirl with the predominant motion occurring at the turbine. The next mode is the critical mode showing large motion at the engine/shafting interface with large LP rotor participation. This mode provides a margin of 23.4 percent over the normal design speed. The effect of the Thomas coupling overhang on critical speeds was determined and is shown in Figure 16. This illustration shows that two modes are involved. For large overhangs the critical mode is a simple cantilever mode of the extended torquemeter, which occurs near the normal design speed. Another mode, which is a coupled LP shaft and output shaft mode, occurs near 15,000 rpm. As the Thomas coupling overhang is reduced, these two modes (eigenvectors) approach the same frequency (eigenvalue). As

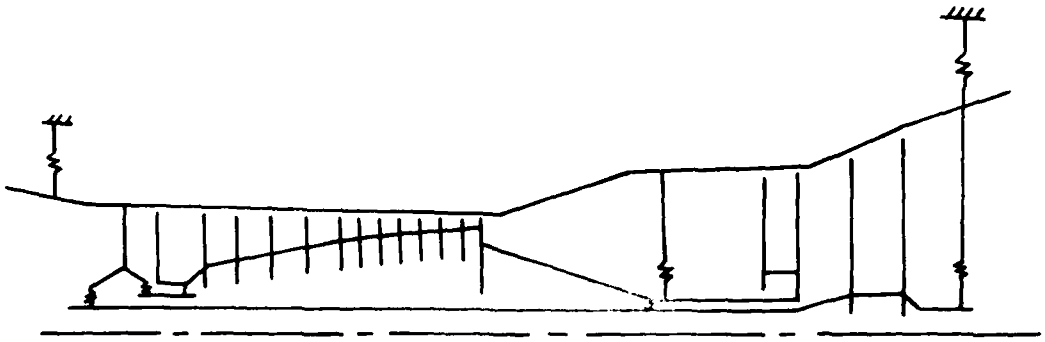


Figure 14. Geometric Representation of 501-M62B Engine Analytical Model.

the overhang is further reduced, the eigenvalues exchange eigenvectors. In this way the modes effectively cross. It can be seen that Mode 2 becomes Mode 1 as the Thomas coupling overhang is reduced from 17 inches to 0.5 inch. Similarly, Mode 1 becomes Mode 2. This means that the first critical speed above design speed cannot be raised above about 14,300 rpm by decreasing overhang alone. Some other design change must also occur.

Over 35 engine/shafting configurations were analyzed, including combinations of two- and three-section shafting designs with original and shortened torquemeter assemblies. The concept of containing the torquemeter within the BV shafting was also considered. The analyses revealed that the only practical solution was a drastic shortening of the torquemeter and housing, in combination with a three-section shafting design from the engine to the mixbox.

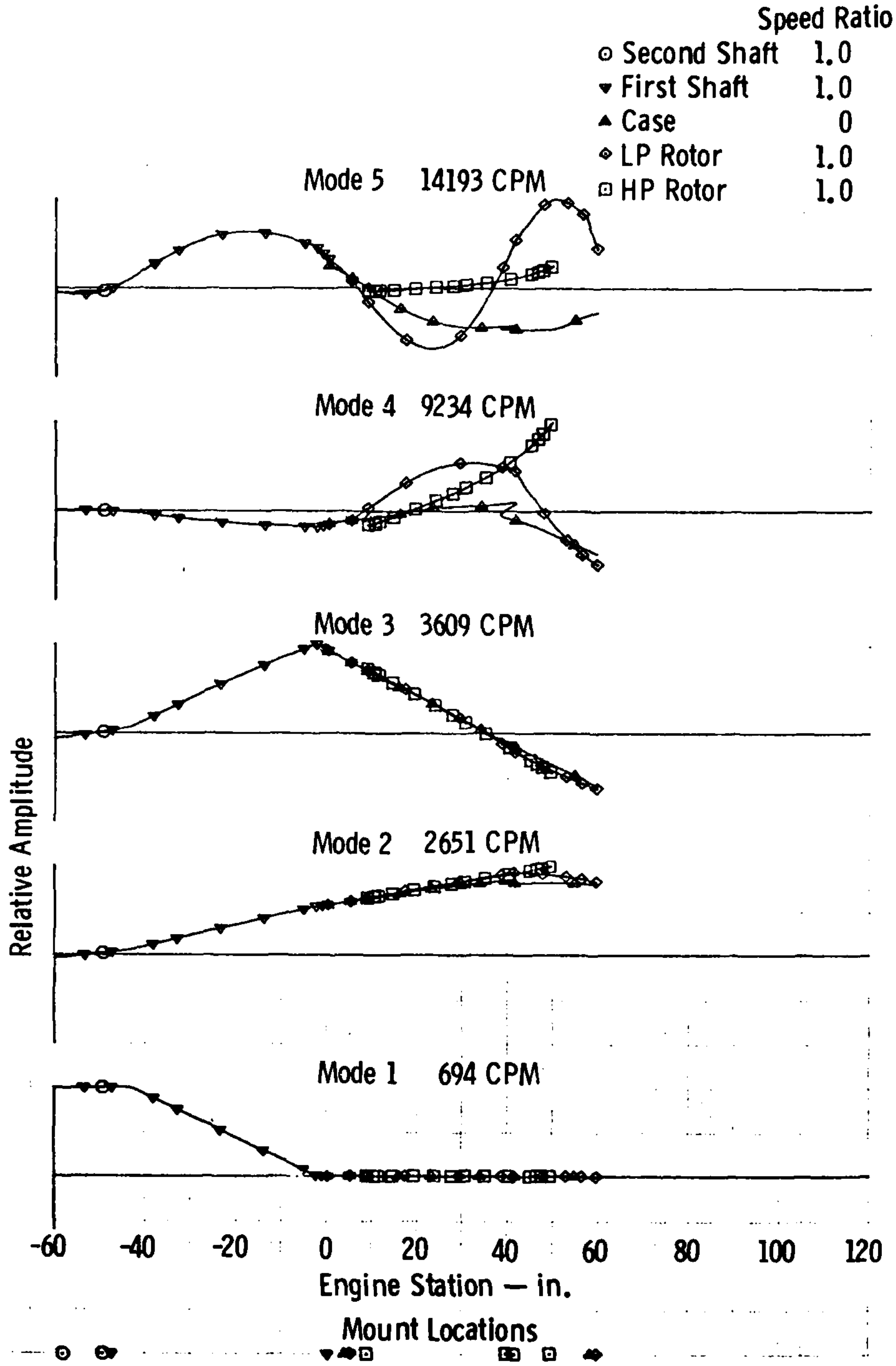


Figure 15. Critical Speeds for 2.4-Inch Thomas Coupling Overhang.

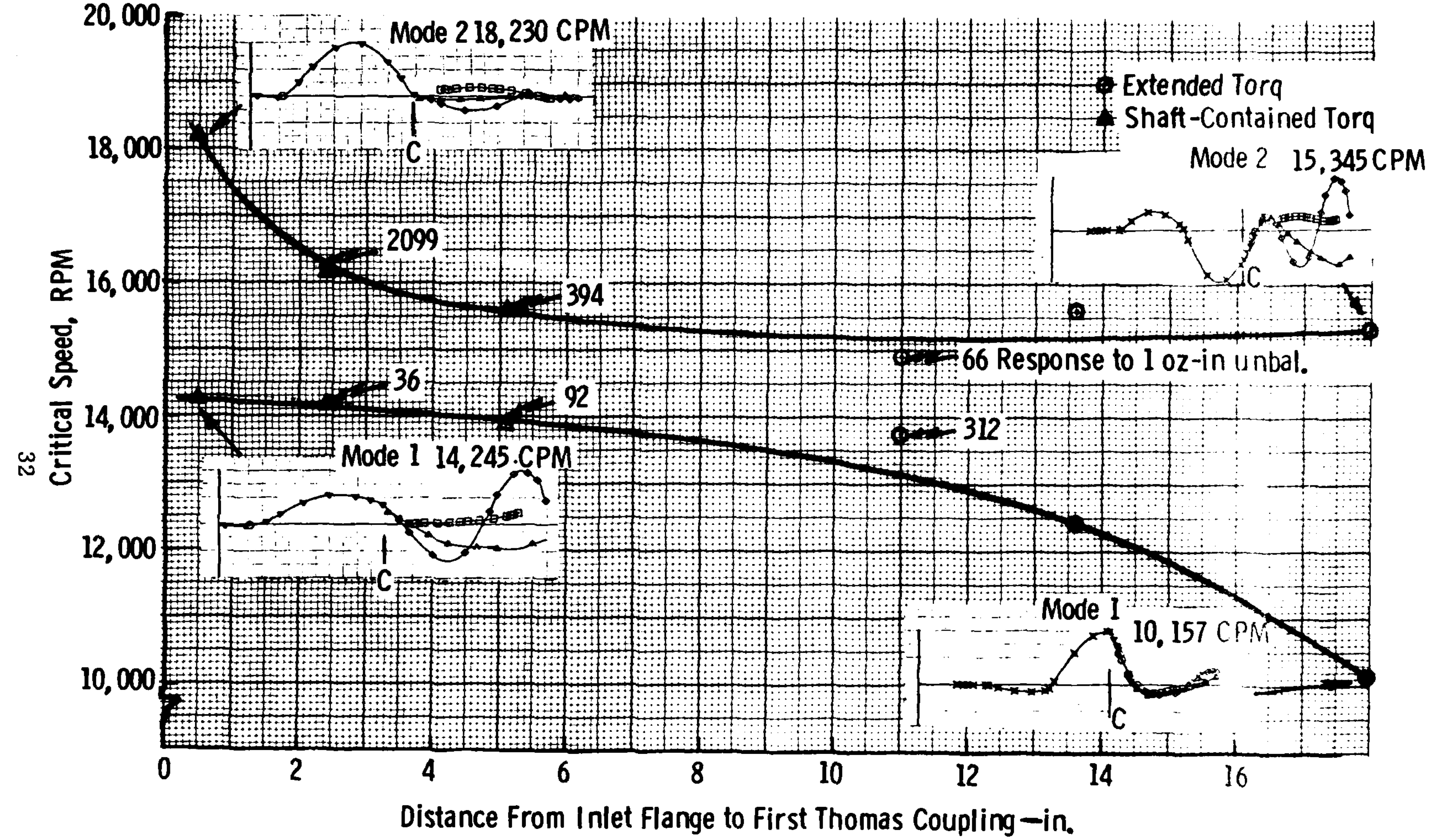


Figure 16. Effect of Thomas Coupling Overhang on Critical Speeds.

RESULTING SOLUTION

The results of the design investigations indicated a shortened torquemeter/three section shafting configuration would provide acceptable vibratory characteristics. An engine design was synthesized to minimize the overhang of the engine/shafting interface and still maintain sufficient torquemeter accuracy. This was accomplished by shortening the torquemeter and containing it within the engine. Figure 17 presents the final 501-M62B design. As can be seen, the Thomas coupling shaft adapter became an integral part of the engine assembly and is overhung 5.274 inches from the engine inlet flange. The BV shafting from the engine to mixbox is similar to the three-section design shown in Figure 12. The sketch of Figure 18 indicates the distribution of shaft lengths. The first section of shaft is 28.085 inches long (center-to-center of the flexible coupling) and is attached to the engine output flange which serves as the engine side of the first flexible coupling adapter.

A coupled engine/shafting analysis of the final configuration was performed. Figure 19 shows the geometric representation of the analytical model. Figure 20 shows the computed critical speeds and associated mode shapes. The first mode above design speed occurs at a critical speed of 14,278 rpm, providing a margin of 24.1 percent. The sensitivity of this mode to an unbalance of 1 ounce-inch at the Thomas coupling nearest the engine is shown in Figure 21. Less than 10 mils of double amplitude displacement is predicted at the resonance, which falls outside the operating range of the rotor system. This configuration is a viable design.

Since the frequency of the first mode above design speed is of such importance, the design/analysis was audited to identify potential weaknesses and to formulate contingency modifications. It was determined that once the shafting and torquemeter designs were fixed, the most influential parameter on determining the frequency of the critical mode was the spring rate of the rear turbine bearing support. The spring rate assumed in the analyses was 500,000 lb/in. A slightly reduced rate could place the critical mode in the operating range of the LP rotor. Therefore, a controlled mechanical flexibility (isolator) was designed for use as a contingency design. Figure 22 shows the effect of employing a spring rate of 100,000 lb/in. The effect has been to reduce the frequency of the 14,278 rpm mode to 8349 cpm without significantly altering the other modes; the mode would be encountered only during starting and shut-down transients. The first mode above the normal LP design speed occurs at 16,958 rpm, an LP margin of 47.4 percent. This same mode has only a 13-percent margin above the HP design speed. However, during the entire testing program of the 501-M62B engine, evidence of the existence of this mode was never discovered. The critical speed occurring at 10,276 rpm is basically an HP turbine mode with significant LP turbine and aft case participation. The mode occurs when considering the HP and casing only but does not occur when considering the LP and casing only. The use of a mechanical flexibility of 100,000 lb/in. provides a viable solution to a potential problem.

During subsequent testing of the 501-M62B engine and engine components, it became clear that the LP turbine rear support rate (without a flexible isolator) was indeed softer than the assumed 500,000 lb/in. An LP turbine resonance was found to occur near the normal design speed. Application of the 100,000 lb/in. isolator reduced the frequency of the resonance and provided acceptable vibratory characteristics. This became the final configuration for the engine delivered to BV for testing on the DSTR.

The results of the combined investigative efforts of DDA and BV were experimentally verified on the DSTR. The engine and DSTR operating times are shown in Table 5. During the 147 hours, 49 minutes, accumulated on the DSTR, not one operating problem was attributed to undesirable shafting/engine rotor dynamics. The engines and shafting proved to be dynamically compatible.

TABLE 5. DSTR Operating Times.

Total test rig time	- 147 hours 49 minutes
Total "on condition" endurance	- 102 hours 54 minutes
Ratio endurance to total time	- 1:1.4
Total number of test rig starts	- 181
Total number of rotor brake stops	- 44

Engine operating time

Engine ident.	Run Times (hr:min) At DDA	At DSTR	Total	DSTR starts
S/N 2	198:45	141:33	340:18	182
S/N 3	18:40	112:54	131:34	133
S/N 4	16:18	135:46	152:04	143
S/N 5	16:15	28:07	44:22	55
Total	249:58	418:20	668:18	513

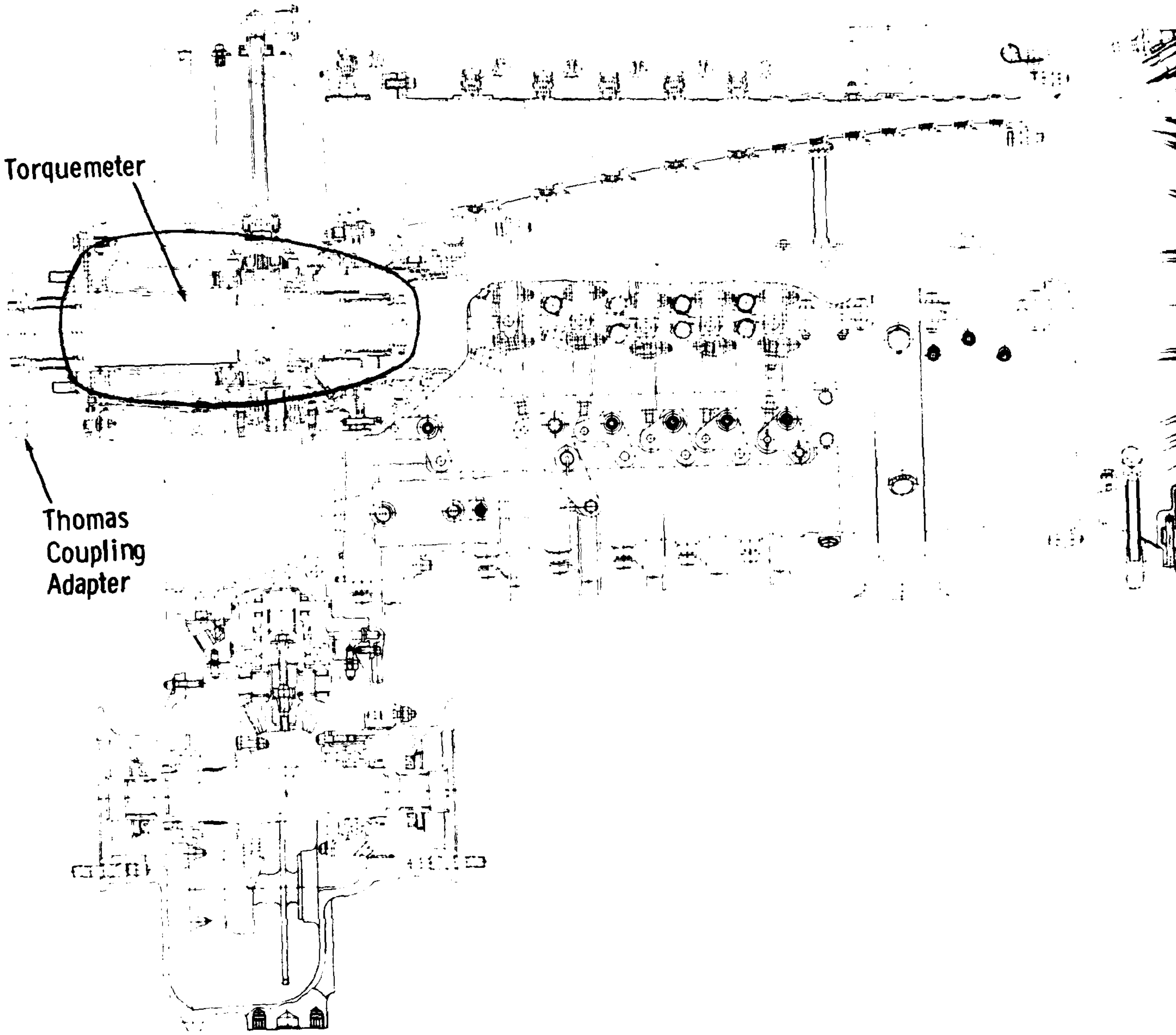
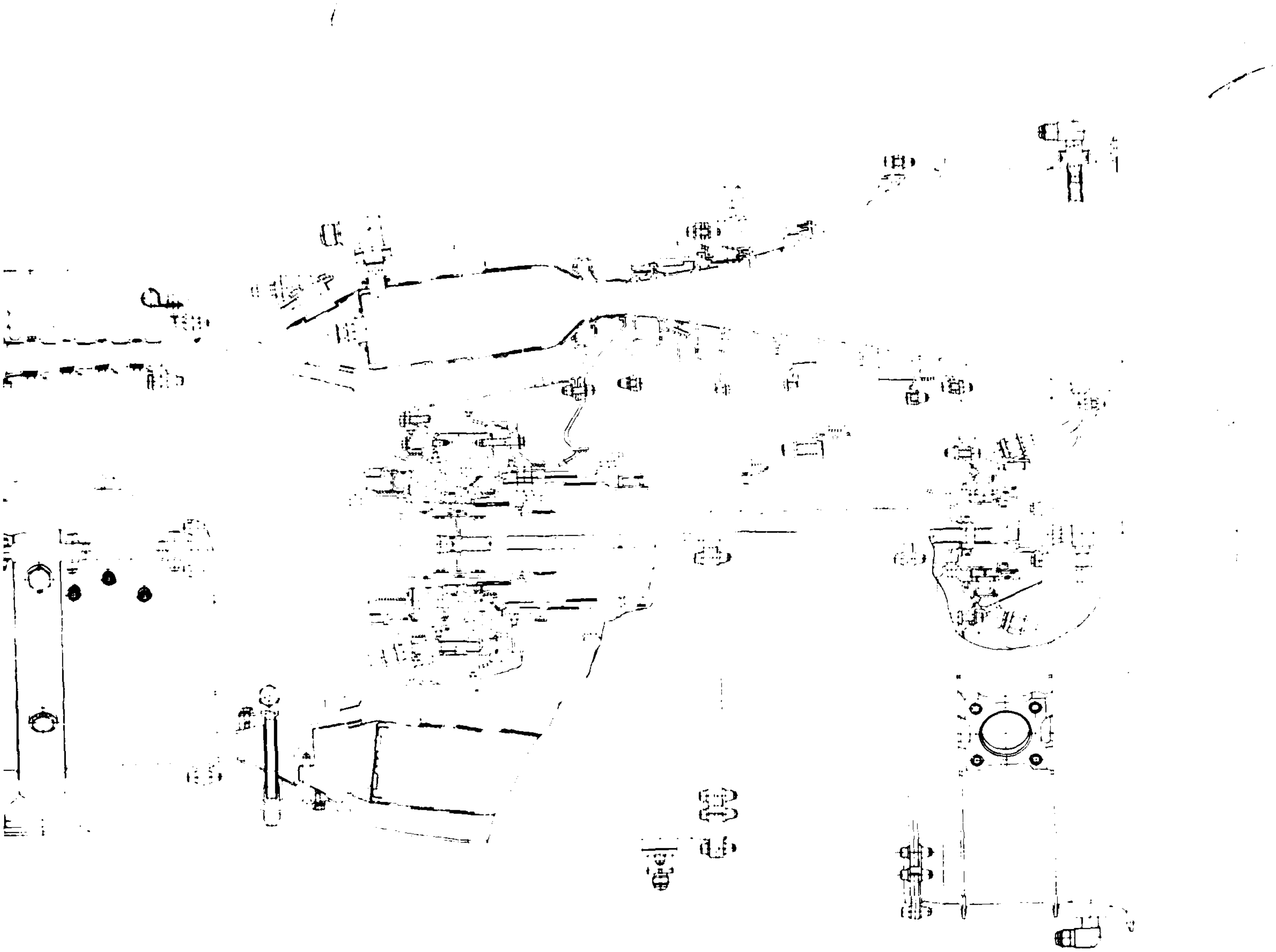


Figure 17. Final 501-M62B Design.



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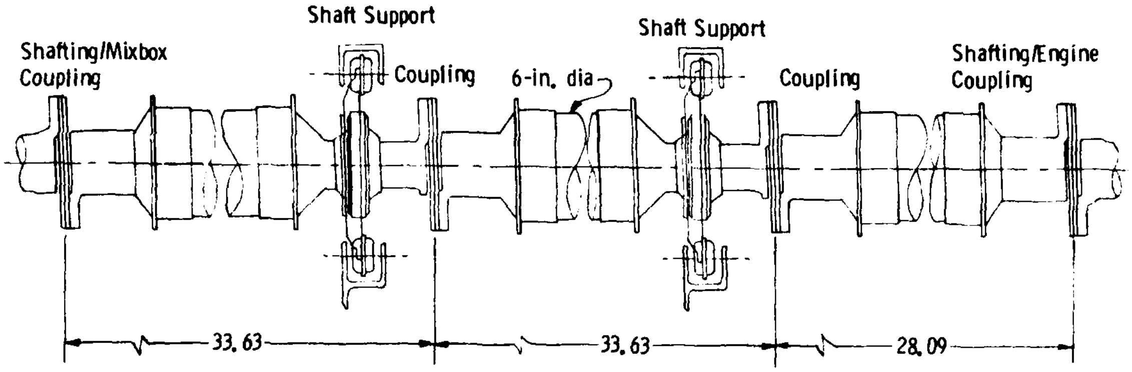

Figure 18. Final Three-Section Shafting Lengths.

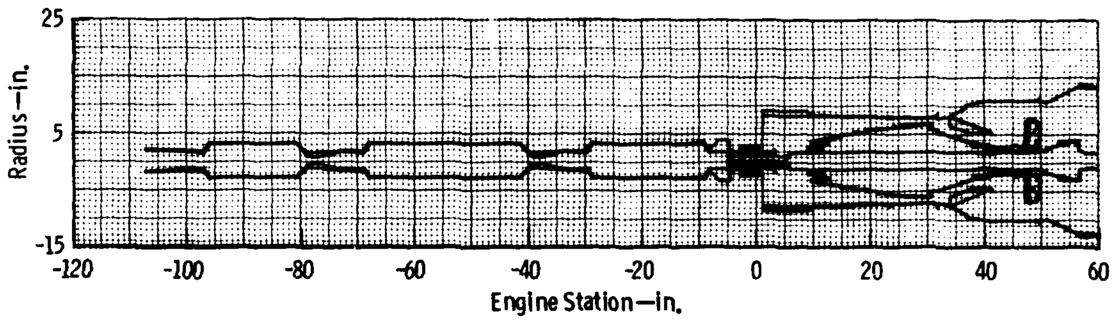

Figure 19. 501-M62B/Shafting System Dynamics (AL-18584)—System Geometric Representation.

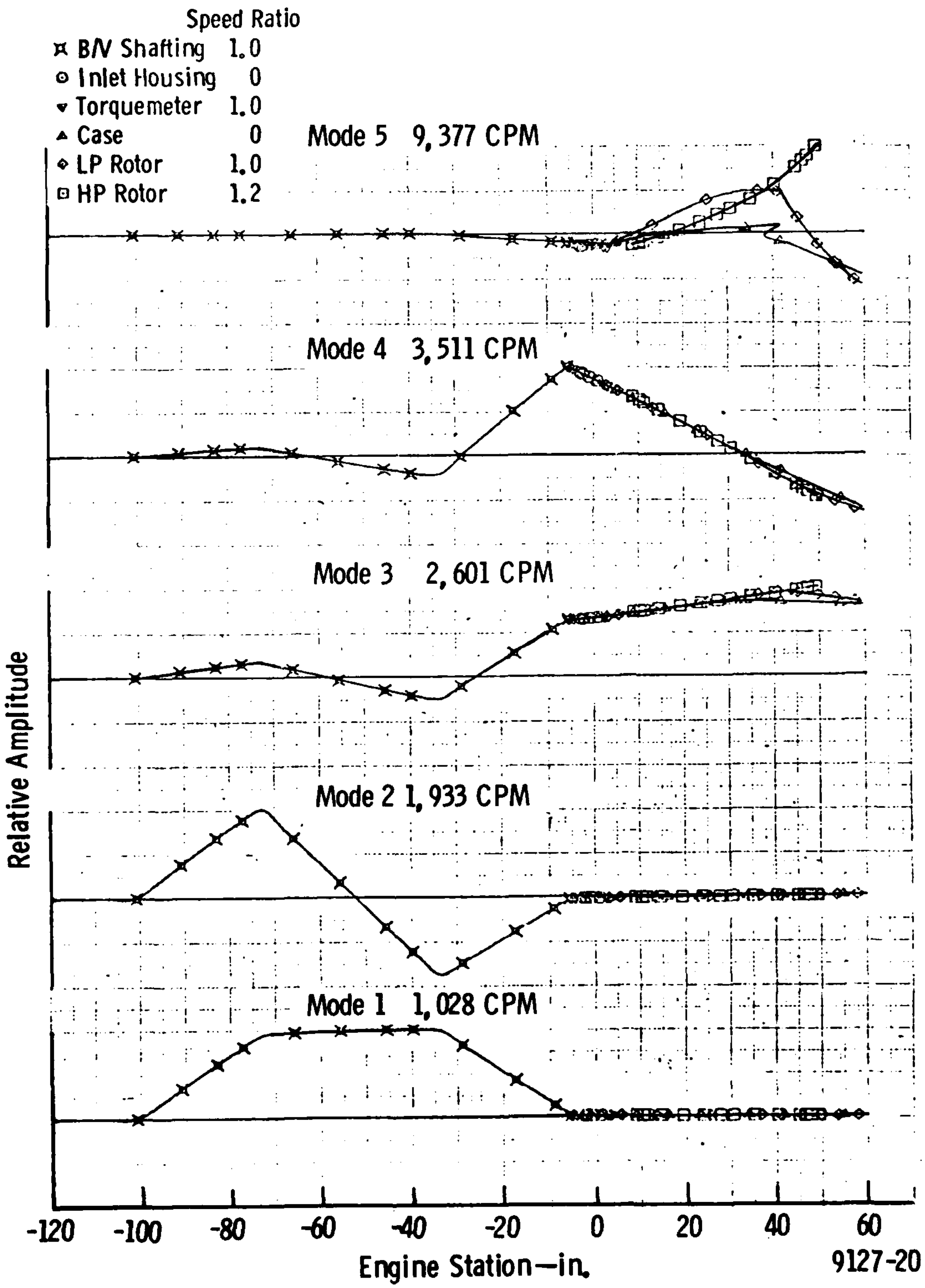

Figure 20. 501-M62B/Shafting System Dynamics (AL-18584)—Nominal Design. (Sheet 1 of 2)

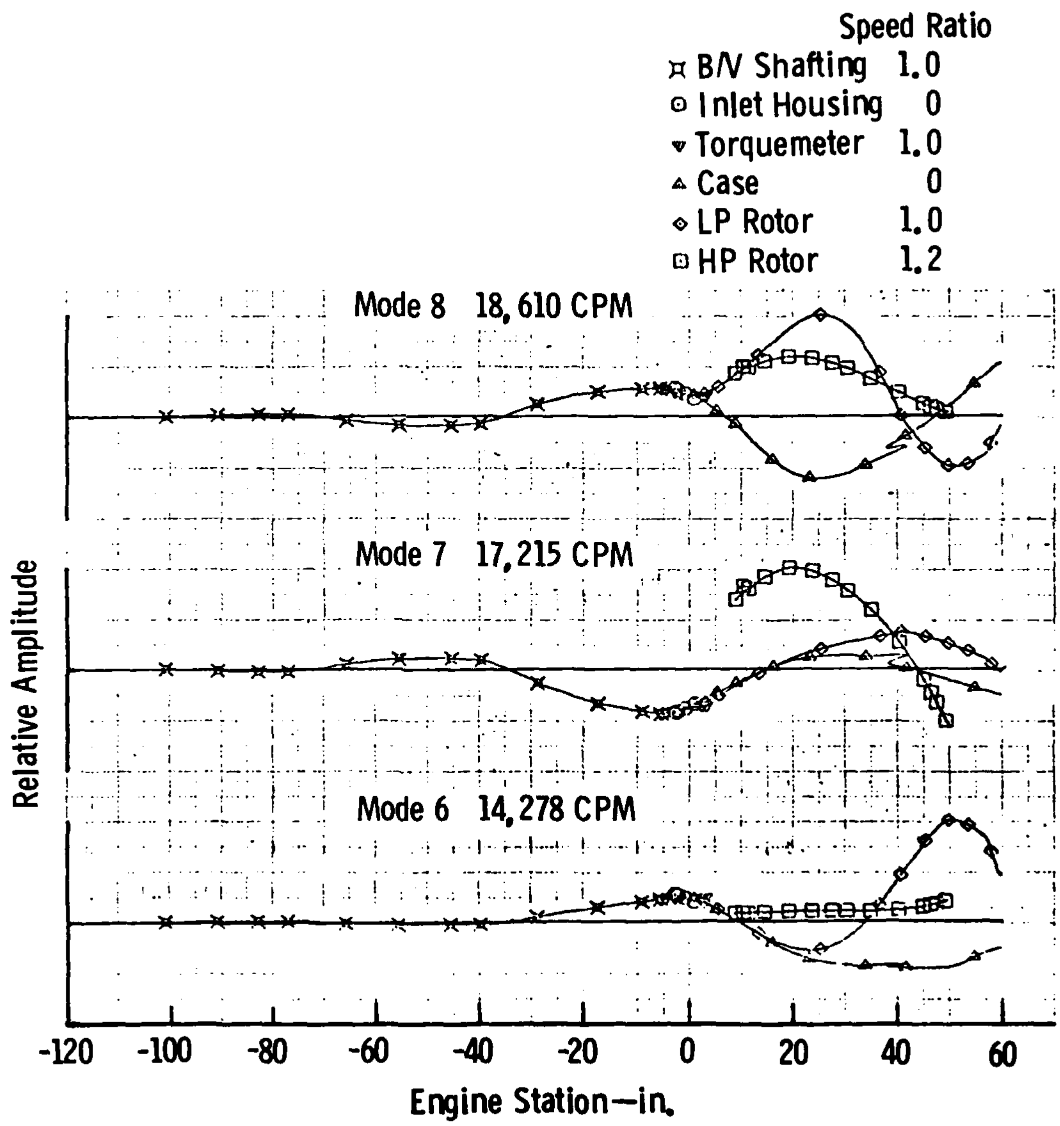

Figure 20. 501-M26B/Shafting System Dynamics (AL-18584)—Nominal Design (Sheet 2 of 2)

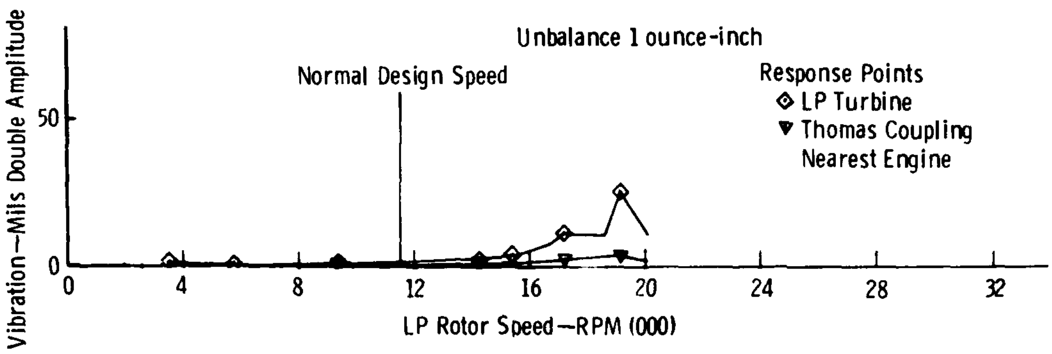

Figure 21. Nominal Design Response to Thomas Coupling Unbalance.

The final design of the coupled engine/shafting system for the DSTR has been presented. Arrival at this design resulted from coordinated design and analysis efforts at DDA and BV. The final design was shown to be free of coupling-related rotor dynamics problems. Testing which followed at the DSTR proved the design to be without excessive rotor dynamic vibratory response. The important effects of the engine LP turbine rear support rate could not have been evaluated without including the complete engine model in the coupled system analysis.

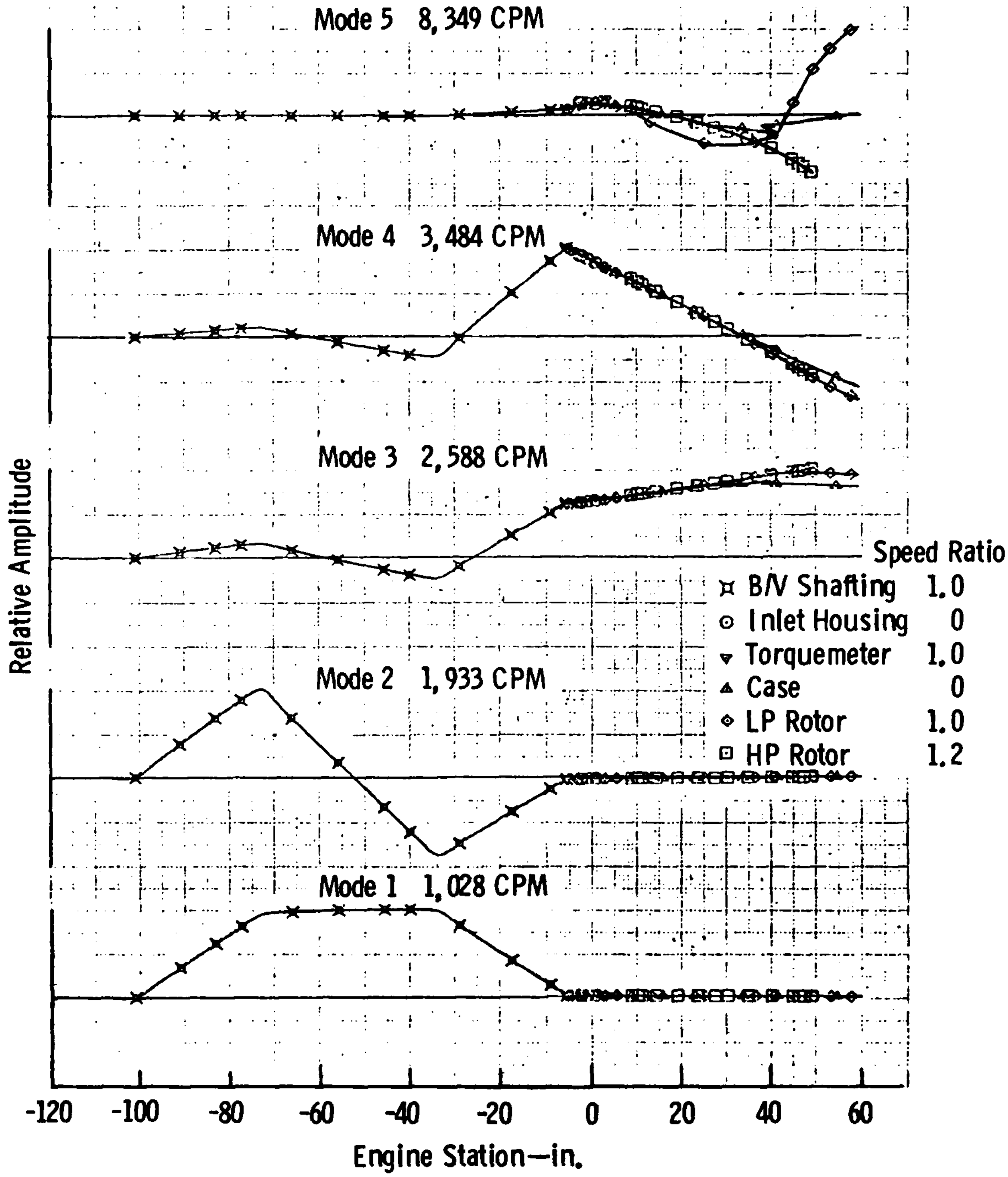

Figure 22. 501-M62B/Shafting System Dynamics (AL-18584)—With Isolator. (Sheet 1 of 2)

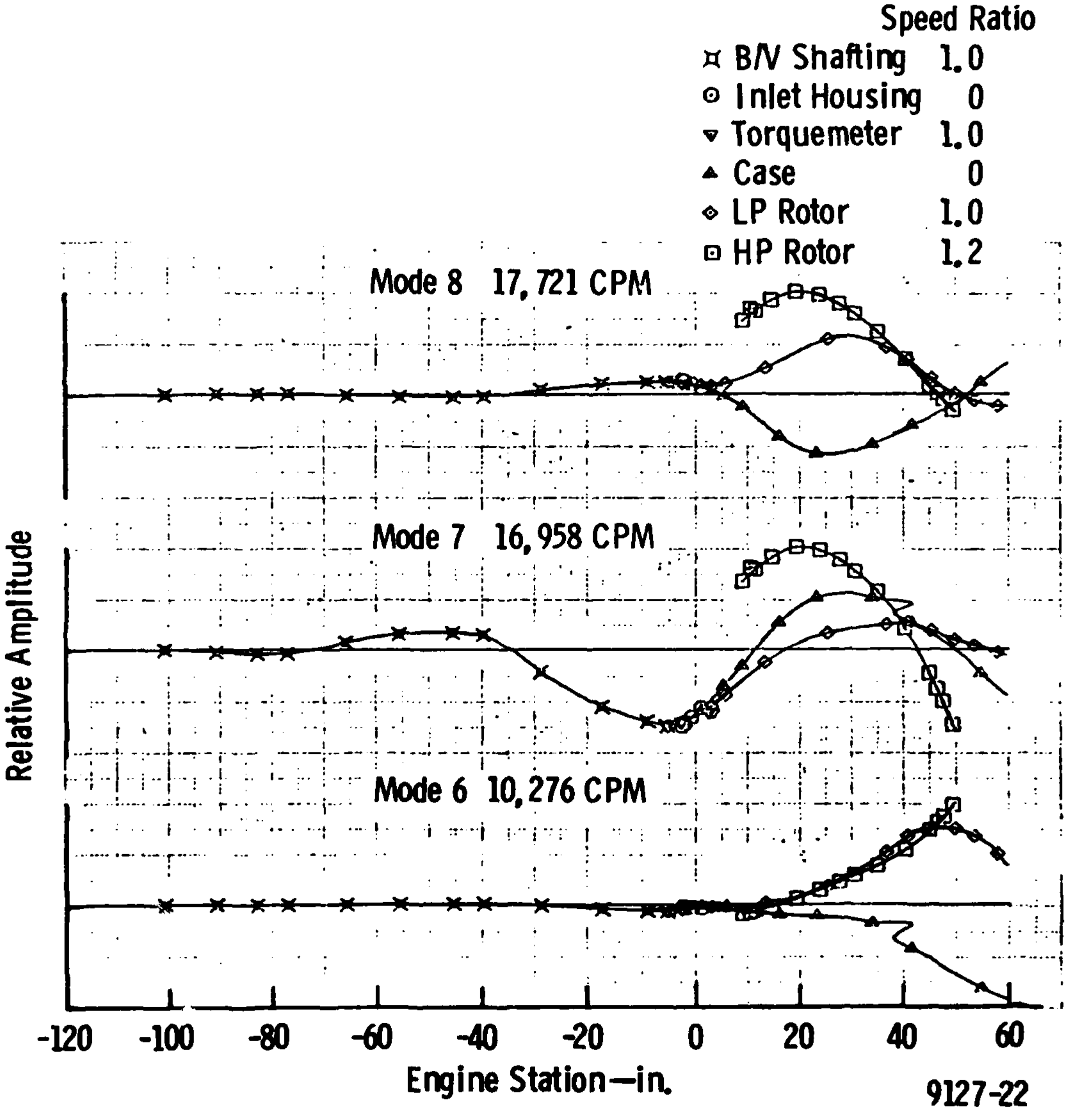

Figure 22. 501-M62B/Shafting System Dynamics (AL-18584)—With Isolator. (Sheet 2 of 2)

SUMMARY

This report has presented a discussion of a classic coupled vibrations problem which had a significant impact on the engine and shafting design for the DSTR. The two major components, the 501-M62B engine and the DSTR engine-to-mixbox drive train, had acceptable vibratory characteristics when considered separately. An early coupled analysis, using information required by Specification MIL-E-8593A to be provided, indicated an acceptable system. However, subsequent analyses with a more detailed engine description showed the coupled system to be unacceptable. A design investigation was conducted to synthesize an acceptable design. A solution could not be found by modifying only one of the major components. Modifications of both the engine and shafting were required to arrive at a viable design. The resulting configuration featured a 6-inch-diameter, three-section engine-to-mixbox drive train, and an engine with shortened torquemeter having minimum extension from the engine inlet flange. The need for a controlled mechanical isolator at the LP turbine rear support was determined at engine testing at DDA. Subsequent testing on the DSTR verified the final drive train configuration.

The evolution of the design process in this program is not unique. However, it is desirable to perform the rigorous coupled analyses prior to committing engine and/or shafting hardware. The loss in manpower, time, and money often accompanying a solution to a problem involving existing hardware can and should be avoided. The information related to the engine/shafting interface required in Specification MIL-E-8593A is often not sufficient to perform a coupled analysis. In the problem presented here, the engine and spring rates (moment and shear) were not sufficient to account for the engine mass effects and certainly could not have been used in predicting the complicated effect on the engine LP turbine mode. A better specification, one which can uncover special problems of the type discussed in this report during the design phase, needs to be formulated for use in future helicopter programs. Such a specification should require a detailed coupled analysis. This analysis must, in general, consider a flexible model of the engine as well as the shafting. The engine and shafting models must be made available in a form which can be readily assimilated.

RECOMMENDATIONS

Based on the results of this effort, it is recommended that an interface specification be written to cover the coupled system analysis of a helicopter engine/shafting drive train. Specifically, this specification should require the engine manufacturer to:

1. Provide an accurate mass-elastic model of the engine including all major subcomponents (casing, rotors, extension shafts, etc).

2. Provide design assistance during any perturbations of the engine design required to synthesize an acceptable drive train design.

The airframe manufacturer should be required to furnish the same information and assistance relative to the airframe shafting. Since the form and content of the data made available may not be the same for each manufacturer, these items must be negotiated. The airframe manufacturer, or the party dictating the helicopter system design, should be given the responsibility of performing the analysis and recommending any changes necessary to synthesize a dynamically acceptable configuration.